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HIGH VELOCITY IMPACT

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I ABSTRACT

A machine was designed and constructed that was
intended to deliver an impact load of high energy
on a tensile specimen at speeds varying from 100-500
ft. per second. Instrumentation, using resistance
type strain gages was set up to measure and record
strain versus time at two points on the sample
during the period of rupture. One specimen was tested
at 57 fps with this arrangement. Two other specimens
were tested without instrumentation at 132 and 189 fps.

Elongation and reduction of area data were
compared with static values. On highest velocity test
an element of the machine failed mechanically and put
an end to testing. Results indicated further
modifications of machine and instrumentation are
necessary before large-scale investigations can be
undertaken. Results of tests on 2S aluminum showed
increasing ductility and reduction of area with
increased speed in region beyond critical speed contrary
to usually accepted theory.

I N D E X

II OBJECT

1.   To develop a method of dynamically loading a
tensile specimen at velocities within the limits
of 100 and 500 fps.

2.   To determine the tensile properties of aluminum
under high velocity impact conditions by means of
an electrical strain gage directly attached to the
specimen in the vicinity of fracture and by comparison
of elongation and reduction of area at various
velocities.

III INTRODUCTION

The ever-mounting demand for speed and more
speed in the rotating and translatory devices of a
modern technology that is reaching for the last
fraction of increased efficiency has, in the last
decade, confronted designers and engineers with the
problem of designing for impact at high velocities.
Admittedly the problem insofar as strength alone is
concerned can be circumvented by liberal application
of very arbitrary safety factors and this, in general,
has been the rule for static structures. There are
certain fields, however, in which the rewards accruing
to a superior design that takes fullest advantage
of the true impact strength of the materials in use
are so rich that the principles of such design methods
can no longer be overlooked. This is most true in
applications where weight is an important factor
and merely fattening up a section cannot replace a
thorough and logical stress analysis. Ordnance
equipment of various types, both projectiles and
launching structures, aircraft elements, blast-proof
designs, and applications in a yet unpredictable atomic
energy technology, might be listed among the fields

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that would be most affected by a complete exposition
of the high velocity impact problem, resulting,
it might reasonably be expected, in designs of
higher efficiency and greater economy.

Prior to about 1936, very little was known
of the behavior of materials under high velocity
loading. At that time H. C. Mann carried out a
series of tests[2] which opened the field and
pointed up the prevailing ignorance of the funda-
mental theory underlying the phenomena observed.
It may be said that the presentation of the subject
as a legitimate analytical branch of materials
theory came with the reading in 1941 of a paper
by Theodore Von Karman before the National Academy
of Science, "On the Propagation of Plastic Deformation
in Solids"[11], in which was introduced an analysis
and a theoretical basis for the behavior of materials
under high velocity impact. Since then, several
investigations have been carried out which have
substantiated in some ways von Karman's proposed
theory, and in its amended form it stands today as
the authority upon which the new study is based.
It does not by itself give the answers. Just as the
theory of elasticity does not solve all static design
problems, the von Karman work is clearly limited by

assumptions and non-ideal conditions. It will, however, serve as a welcome foundation upon which investigators can base future research projects into more specific questions of the field.

SUMMARY OF THEORY[9]

Within the elastic limit of a material, the stress in a semi-infinite bar with one end fixed whose other end is instantly subjected to any velocity V_0 can be computed from the following relationship:

$$e = V_0 t = \epsilon.1 = \epsilon.c.t$$

where e = total strain

V_0 = velocity of end of bar

t = time after velocity began

ϵ = unit strain in bar

1 = length of strained part of bar = distance stress wave has progress in time t.

c = velocity of stress wave = $\sqrt{\dfrac{E}{\rho}}$ = $\sqrt{\dfrac{\text{Young's Modulus}}{\text{Mass density of material}}}$

so $V_0 = \epsilon.c$

$\epsilon = \dfrac{V_0}{c}$

and stress in part of bar being strained = $\sigma = \epsilon.E = \dfrac{V_0}{c}.E$

Theodore von Kármán about 1942 undertook the extension
of this method of rational analysis into the plastic
range by introducing the variable slope $\dfrac{\partial \sigma}{\partial \epsilon}$ as a
substitute for E to include the plastic region of the
stress strain curve. It then follows that the velocity
of the plastic stress wave becomes,

$$V_\epsilon = \sqrt{\dfrac{\partial \sigma}{\partial \epsilon}\Big/\rho}$$

So for the same case as above in both elastic and
plastic ranges

$$e = \int_0^\epsilon V_\epsilon \cdot t \cdot d\epsilon \qquad \text{by analogy}$$

so

$$V_0 t = t \cdot \int_0^\epsilon V_\epsilon \cdot d\epsilon$$

or

$$V_0 = \int_0^\epsilon V_\epsilon \cdot d\epsilon$$

Then if ϵ is assigned the value corresponding to
the strain at which necking begins, ϵm, the
required impact velocity to cause abrupt failure
at the moving end may be indicated.

$$V_0 = \int_0^{\epsilon_m} V_\epsilon \cdot d\epsilon$$

Given the static stress strain curve for the material,
the value of the integral can be computed. This

value V_0 is then known as the "critical velocity"
or that velocity at which failure will occur at
the extreme end of the bar and no plastic
deformation will take place beyond it.

This failure is supposedly characterized by
low absorption of energy and small total elongation.
It is important to note that although V_0 as applied
to the end of the bar is assumed constant, it may
be attained in any manner and still cause failure
though, in the progressive case, plastic strains
may be found in regions removed from the end.

The above discussion although necessarily
abbreviated, presents the principal points of the
generally accepted theory governing high velocity
impact. This paper presents in the discussion of
results, an application of the critical velocity
concept to the material being tested.

In selecting a topic for thesis work, the
investigators decided that it would be a step
forward to have available at the Institute, a
machine with which high velocity impact investigations
might be carried on by future workers. There are
very few such machines in existence. It was hoped
that work would be completed in time to allow

sample tests of aluminum specimens to
determine what information might be obtained from
them. The proper and complete study of high
velocity impact must eventually include point
to point determination of stress and strain in
the elastic and plastic ranges of samples subjected
to tensile, compressive, bending, and torsional
forces. This would be a formidable undertaking.
It was decided to restrict the project to
tensile tests applied to conventional tensile
specimens, to measure strain at several points
on the specimen, and to confine instrumentation
to that required for recording strain versus time.
This implies neglecting energy of fracture, a
characteristic considered most important in the
conventional Charpy and Izod tests. However,
a preliminary study of work performed on the
subject indicates beyond reasonable doubt that
at relatively low velocities the energy absorbed
by the specimen approaches zero. The work
contemplated being beyond this velocity range,
it was decided to neglect energy as indicative of
impact properties and to concentrate efforts upon
a study of ductility, stress, and strain distribution
in the specimen.

The material used for all of the tests was
2S aluminum. It was received as $\frac{1}{2}$" diameter
bar stock and was subsequently machined to the
shape shown in Fig. (1). The specimens shown
with the longer ends were used for static tests
and those with the short ends for the dynamic tests.

The static tests were made using a Riehle,
(30,000 lb. capacity), loading machine for measuring
the load and an A-5 electric strain gage for
measuring the strain. A stress vs. strain curve
was plotted and the physical characteristics of
the material was computed as shown in Figure (2).
The material was found to have the following
characteristics under static conditions:

1. Yield strength at permanent set of .0002$^{in}/_{in}$ = 12,700 psi

2. Ultimate strength of   18,000 psi

3. % Elongation of   20%

4. % Reduction in area of 61.3%

5. Modulus of elasticity of 9.55×10^{6}

It was first thought that the testing procedure
could best be applied to low carbon steel since it
is in such common use and so readily obtainable in
most any shape. It was soon learned that due to the

relatively high strength of steel it was necessary
to make the testing machine much larger than it
would have to be if it were used on some material
with lower strength characteristics.

Plastics were then considered as a material
that might be tested, but it was learned that
the data that could be obtained from strain gages
would be limited. This is true since the electric
strain gage is only capable of indicating strain up
to a certain limit, at which time it will either
break or the bond between the paper and material will
separate. With this in mind, it will be simple
to see that when a material such as plastic with a
very low modulus of elasticity is loaded it would be
possible to reach the limit of the strain gage and
still be well below the elastic limit of the material.
For this reason plastics were not used as a material
for the tests because it seemed advisable that the
results of the tests should give data pertaining to
the behavior of the material up to the yield and
as far beyond as the strain gage would permit.

It was felt that use of 2S aluminum represented a
compromise between that of steel and plastic, in
that it would yield the desired type of data without
requiring a large piece of testing apparatus.

V APPARATUS

A. IMPACT MACHINE

The selection of the type of apparatus used
for dynamically loading the specimen was made by
carefully considering several methods. The decision
was first made that the machine should possess
the following desirable characteristics.

1. Adequate speed control within the limits
   of 100 fps and 500 fps.

2. Capable of loading specimen in tension.
   It was believed that loading in this manner
   should yield results that could easily be
   compared to well established data on material
   characteristics under static loading conditions.

3. Simple in design. This was necessary since
   shop facilities would not permit the design
   of complicated parts requiring the use of
   high precision equipment.

4. Light in weight and requiring little space.
   This requirement was necessary in order that
   the machine may be easily transported and
   assembled within the confines of the laboratory.

Some of the loading methods under consideration were
as follows:

a) Projectile propelled by an explosive charge.

b) Projectile propelled by compressed air as shown
   in Figure (2).

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c) Projectile propelled by a rotating wheel
as shown in figure (3).

d) Forked hammer released from a rotating
member as shown in sketch (4).

In general the type of loading which depends
upon a projectile for the breaking energy does not
lend itself well to tension loading since the impact
force must be transmitted in a direction opposite to
and along the same path as the moving projectile.
This has the disadvantage of requiring a complicated
coupling between the point of direct impact and the
end of the specimen, and furthermore, it introduces the
possibility of eccentric loading on the specimen.
Another chief disadvantage of the projectile is that
it must be so proportioned in size that it is large
enough to possess adequate breaking energy at the lower
velocities and at the same time be small enough to
permit its being propelled to the higher velocities using
the same apparatus with limited power facilities.
This suggests that a different projectile must be used for
every range of velocity, all of which further com-
plicates the apparatus. A method for the firing of
a projectile at controlled speed is a difficult
problem when viewing it from the standpoint of sim-
plicity in construction. Speed control within any

degree of accuracy using an explosive charge is
almost impossible to obtain and, therefore,
this method was discarded as a possible choice.

The method using compressed air was considered
in detail and it was found that the machine would be
too large and would require too much in the way of
precise machine work in connection with the
fabrication of the cylinder and piston air seal.

The method using a projectile propelled from a
rotating wheel was discarded because the ball
projectile had to be approximately 1½ inches in
diameter to obtain the desired striking energy which
requires an excessively large specimen, wheel,
race and driving motor.

It was finally decided that the rotating
device using a forked hammer, (as shown in Figure (4))
was the method of all those considered which best
suited the purpose. Detailed design calculations
indicated that it could be made in one small
compact unit, it could be made with a minimum of
machine tools, and possessed the distinguishing
features of having very good speed control and being
capable of loading a specimen in tension. A simple
and direct couple between the point of impact and

the end of the specimen could easily be devised
by using a rectangular block hereafter called a tup,
which is attached directly to the loaded end of the
specimen and which receives the impact blow from
the forked hammer. This is illustrated in Figure (4).

The design of the machine which was finally used
was based on the premise that it would be driven by
an electric, direct current motor with a wide range
of speed control. A motor of this type was available
which had a speed range between 100 RPM and 1800 RPM
so it was decided that it could be used. This choice
affected the diameter of the rotating member and,
therefore, a diameter of 3 feet was used. This made
the distance between the center of rotation and the
center of impact equal to 18 inches. The linear
relationship between the angular velocity of the
member and the striking velocity of hammer was then;
Revolutions per Minute x 0.1571 = fps. Using a two
to one pulley connection between the motor and the
rotating member allowed a possible velocity range
of 39.3 fps to 560 fps.

The apparatus shown in photographs A to E was
devised as the best solution to the problem from the
standpoint of simplicity in operation and construction.
It consisted to a 3"O.D. x ½" wall steel tube through

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which the striking hammer could slide. The
mechanism for releasing the hammer consisted of
a ring equipped with slots which engaged a pin
mounted in the hammer as shown in photographs D and E.
This ring was made with a projecting lug which engaged
a tripping plunger as shown in photograph D. When
the contact between the ring lug and the trip plunger
was made, the rotating movement of the tube forced
the ring to rotate about the tube and thus release
the hammer. The hammer was to be propelled by
centrifugal force along the axis of the tube until
it was stopped by the pin bearing against the end
of the slot in the tube. The forked end of the
hammer was then in a position to clear the specimen
and to engage the tup, thus imparting the impact loading
that was desired. The opposite end of the tube was
provided with a means of balancing the entire rotating
assembly. A movable weight was used for this purpose
which was actuated by screw threads and provided with
a lock screw. The tripping plunger was spring loaded
and was actuated by a lever as shown in photograph (0).

The problem of how the specimen could best be
held in a position for striking without creating any
interference with the path of the hammer was solved

by placing that end not struck by the hammer
into a slot as shown in photograph F. The
connection between the tup and the specimen was
made as shown in photograph F in order that the
striking force would be transmitted from the tup
to the specimen uniformly, thus eliminating any
possibility of specimen fracture at that point.

It seemed advisable to have a small initial
load on the specimen in order that it might be
properly seated in the holder slot and in the tup.
This was accomplished by means of a spring loaded
bell crank on either side of the tup as shown in
photographs (3) and (F).

The rotating tube was keyed to a shaft which
was mounted in ball bearings and coupled to a 1 1/4 H.P.;
110 V; 200 to 1800 RPM D.C. motor equipped with
a movable armature speed control. The driving couple
between the armature and the shaft consisted of a
"V" belt drive with a pulley ratio of 2 to 1 from
the motor to the shaft. The direction of this pulley
ratio was reversed at the lower speeds.

After the machine, as originally designed,
had been constructed it was then subjected to a series
of tests to determine just how it would stand up
under operating conditions.

The machine was first tested to investigate
all possibilities of misalignment and destructive
vibrations. This was done by first driving the
machine very slowly and then gradually increasing
the speed up to 1000 RPM, taking notice of the
machine's response to an increase in speed and any
irregularities that occured. This test proved
that the machine was properly aligned and that the
rotating member was balanced well enough to reduce
troublesome vibrations.

The second group of tests was made in order to
determine how the tripping mechanism and the hammer
would stand up under operation. This test was re-
peated four times at low speed ranging between 250 RPM
and 400 RPM. There were no specimens fractured during
these tests. These tests clearly indicated that one
mechanical weakness is inherent in this type of
machine; namely that the pin holding the hammer in
place in the tube slot must be made quite large and
be made from a high grade tool steel. The pin that
was used was made from 3/8" diameter drill rod, but
was subject to bending when the hammer was tripped
at a speed of 360 RPM. It was also learned that the
ring which holds the pin in place and which trips
the hammer was not able to withstand the impact blow
imparted to it when the tripping plunger was released.

It was somewhat distorted in the area around the
end of the slot where the ring had driven itself
against the pin. The screw shown in photograph (D) was
placed in the tube in such a manner that it would
bear against the tapered edge of the ring. It was
believed that the ring would wedge itself between
the screw and the welded shoulder on the tube and
thus prevent it from turning past the point where the
ring could be struck by the pin. This screw was to
further serve the purpose of securing the ring in a
position after impact in order that it wouldn't spin
around and engage the trip plunger again.

After construction of the machine had been
completed, it had virtually the same form as described
above with only slight modifications. The hammer and pin
were rebuilt in such a manner that the weight of the
hammer was reduced by about 25% and the cross sectional
area of the pin was increased by 37%. The pin was
made from SAE 4150 steel, was soaked for 2 hours at
1575°F, quenched in water and tempered at 800°F
for two hours. The hammer and tup were made from
SAE 1045 soaked for 2 hours at 1575°F, quenched
in oil and tempered at 400°F for two hours. The final
shape of the hammer and pin was as shown in Figure (5).
The ring was not rebuilt because it could be made to

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function properly even though it was somewhat
distorted as a result of the first series of
trial runs.   The machine was then ready for the
final test with the specimen in place, the ~~gage~~ strain
gages mounted, and connected to the related
instruments.

B. **INSTRUMENTATION** Figure (7)

The problem of instrumentation presented
was to devise a method of recording the strain-
time relationship during impact at several
points on the specimen. Electrical strain
gages are the only gages available with the
instantaneous response required to accurately
indicate these strain changes that occur in
intervals of time measured in micro-seconds.
(For example, at 800 fps. a moderate loading
rate, assuming instantaneous velocity attained
by the end of the specimen with a 1" gage length,
it would require 0.37 micro-seconds for the
aluminum sample to exceed its yield strain.
The actual time required is of course actually
longer, but how much longer no one knows.)
Of the various types of electric gages in use
the resistance type was selected for its com-
pactness, simplicity, expendability, and avail-
ability. The type C8 gage was selected as the
shortest (1/8") gage length commercially obtainable
suited to dynamic work. A six channel General
Electric strain gage amplifier received the 1000
cycle gage signal through the unbalanced voltage

of a Wheatstone bridge. A four stage amplifier
increased the signal, rectified it and fed it into a
six channel General Electric Oscillograph type
FW-1C-BZ with high speed camera attached to record
the galvanometer traces. The galvanometers
were single element 6S type with a frequency
response of about 5000 cycles. The magazine type
film holder was used to take advantage of the
highest possible film speed, 1270 MPW, which
corresponded to 1270 fpm or 254 inches/sec. On
such a time scale each inch corresponded to 3.94
milliseconds. A Hewlett-Packard Model 202 D
Audio Amplifier was used to feed a timing wave
into the galvanometer at 400 cycles. The set-up
as finally used is shown in Figure (7) and
photograph (K).

Automatic synchronization of camera exposure
and specimen rupture was a problem which was not
solved and was avoided by manually opening the
camera shutter just prior to applying the impact
load and closing it immediately thereafter.

Careful tuning of bridge circuits was important
and found to be necessary every few minutes as the
bridges drifted off balance.

Each gage circuit was calibrated electrically
by placing a known resistance in parallel with
the strain gage and noting the displacement of the
galvanometer beam.

A mechanical steam-turbine tachometer was
used to measure rotational speed of the shaft
upon which the rotating beam was mounted for
computing striking velocities.

One important innovation in the instrumentation
was the application of the strain gages directly
to the sample at the forward and rear ends of the
reduced section (see photograph (F)). It had
been the custom in the past to mount the gages on
an adjacent section which transmitted the impact
load to the specimen. This practice saved strain
gages and did indicate the load being applied to
the end of the specimen, but it gave no indication
of the strain and stress distribution during impact
in the specimen itself in terms of position relative
to point of load application. It was hoped that
direct measurement of strain-time curves of points
on the specimen would provide a basis for analysis
of non-ideal impact. A third C-6 gage was mounted
on the holder of the fixed end of the specimen in an
effort to determine correlation of readings on this
and the direct mounted gages.

1. The strain gages were cemented to the
sample at the desired points and, after drying,
were mounted in the tup and fixed holder.

2. The gage circuits were completed using
soldered connections to shielded leads, taking
care that leads were clear of hammer.

3. The bridge circuits were balanced with amplifier
phase and amplitude adjustments starting at lowest
attenuator setting and working up to #10 gain
setting so that minimum equilibrium current was
flowing.

4. All gage circuits were calibrated with known
resistance in terms of micro-inches/inch per mm.
of galvanometer beam deflection. Used 5 ohm
galvanometer input resistance.

6. With camera motor up to speed the timing wave
was recorded by use of transient shutter control.

7. Impact machine was started and brought up to
desired speed of 360 RPM as indicated by tachometer
by adjusting armature speed control on motor. On
this run 80 VDC was used to energize the motor from
a motor-generator converter set.

8. Lamp voltage on oscillograph was set to maximum

value, camera shutter opened manually, trigger

release on machine was tripped to initiate

impact blow, and camera shutter was closed.

This was done in succession as rapidly as

possible.

9. Lamp voltage reduced. Motors stopped.

10. Gage length and reduced diameter measured.

Temperature of room noted and film record developed

and printed.

The behavior of the loading machine
during all of the tests was observed and
the following mechanical weaknesses were
found to exist:

1. The pin which is mounted in the hammer
and which slides along the slot in the tube
was subject to a brittle fracture close to
the center. This failure took place at the
highest testing velocity of 189 fps. The
fracture took place at a point where a small
quench crack existed. It is believed that
this weakness was largely a result of fully
hardening the pin.

2. The tripping ring was subject to distortion
during every test. This weakness is plainly
a result of too soft a grade of steel and
poor design. The stop that was provided was
not adequate nor was the ring made to sufficiently
large dimensions. The very nature of the ring's
function makes a rational design exceedingly
difficult.

3. The location of the tripping plunger was

such that the hammer was released in such a
position after tripping that it struck the
top of the tup before making a complete revolution.
Evidence that this occurred could be seen when
the specimen was inspected, since large marks
were found in the top end where the tup had gouged
into the specimen as a result of an irregular
loading condition. This is also evidenced in
the photographic oscillograph trace (photograph J),
by the small undulating signal which diminishes
in magnitude as if the specimen were struck and
allowed to damp out its own vibration. It will be
noted that this trace is the same on both channel
#1 and #2 which received the signal from the
gages on the specimen. This irregularity was a
result of poor design since the plunger could well
have been placed beyond the top to avoid this difficulty.
d. The assembly which was subject to rotation at
high velocity caused considerable noise due to
wind resistance and it was noticed that the motor's
speed control wasn't very sensitive at the higher
speeds due to this heavy load imparted to it. It
is believed that the lug on the tripping ring was
largely responsible for the wind friction encountered.

The ring lag could well have been made a little
shorter.

5.   The top-end of the specimen was subject to
some bending at the velocity of 181 fps mostly
due to the blow it received when it was caught
by the box of cotton waste and rags. The box
about 6 inches deep, wasn't deep enough to
cushion the blow properly.

6.   The ends of the slots in both the tube and
the hammer were subject to a small amount of
upset as a result of the impact received when
the hammer was thrown out. This could have
been minimized by providing a cushion at the end
of the slot.

7.   The bearing support channel to which the top
was attached was subjected to more lateral thrust
than was anticipated due to the ring movement
about the tube. The channel was forced outward
enough to cause some clearance between the
shoulders on the driving shaft and the bearing race.
This is serious because the tube assembly might
move far enough in a lateral direction to cause the
specimen holder to be struck by the hammer.

8.   Elimination of random pick-up in the circuit
was not satisfactory as evidenced by the 000 cycle
signal impressed on the recorded signals (photo-
graph I). Fully shielded and grounded leads were

used and all instrument chassis were grounded,
however, it is felt that the prepared plug-in
leads used were excessively long.

9. The manual tripping of the camera shutter is
not a satisfactory method of control since it
caused over-exposure of the zero and open circuit
lines on the record. It was necessary, due to
inability to devise a satisfactory synchronizing
method for camera exposure in the time allowed.

10. Under-voltage on the lamp in the oscillograph
may have been the cause of the faint rupture track
obtained on the film. This was due to a shorted
rheostat control on the oscillograph for which
no spare was available.

11. The timing wave failed to appear on the film
record due to the width of the film being less than
anticipated. It was believed that the film would
record the full length of the visual screen, about
9 inches, whereas it actually extended only over
5 3/4 inches at the left end of the screen.

VIII DATA

Sample A - STATIC TEST

As Gage Factor = 2.01   Sample Dia. = 0.252"

Load	Gage		Total	(psi)
0	5-722			0
50	825	103	103	1000
100	1033	108	211	2000
150	1145	112	323	3000
200	1252	107	430	4000
250	1362	110	540	5000
275	1412	50	590	5500
300	1463	51	641	6000
325	1518	55	696	6500
350	1570	52	748	7000
375	1624	54	802	7500
400	1680	56	858	8000
425	1736	56	914	8500
450	1797	61	975	9000
475	1860	63	1038	9500
500	1925	65	1103	10000
525	6-1001	76	1179	10500
550	1073	72	1251	11000
575	1151	78	1329	11500
600	1238	87	1416	12000
625	1330	92	1508	12500
650	1438	108	1616	13000
675	1540	102	1718	13500
700	1676	136	1854	14000
725	1839	163	2017	14500
750	7-1038	199	2216	15000
775	1288	250	2466	15500
800	1650	362	2828	16000
825	8-1218	568	3396	16500
850	10- 180	962	4358	17000
875	-1950	1770	6128	17500
900	46-1400	9450	15578	18000
916 A 10-2000+		11000+	26578+	18000
925	Failure	-	-	

Gage Length   Initial = 1.000"
               Final   = 1.303"

Diam.   Initial = 0.252"
               Final   = 0.109"

ELECTRICAL CALIBRATION:

R (Calibrating Resistance) = 161,000

$$\epsilon_c = \frac{R_g}{GF\ (R_c + R_g)} \qquad (12)$$

for C-B gages

GF = 2.86

R_g = 500

$$\epsilon_c = \frac{500}{2.86\ (161,500)} = 1082\ \text{micro-in./in.}$$

for C-B gage

GF = 3.26

R_g = 362

$$\epsilon_c = \frac{362}{3.26\ (161,362)} = 671\ \text{micro-in./in.}$$

Channel	Amplifier Attenuator	Galvanometer Resistance (Ohms)	Scale (mm)	
			a	b
A	#10	5	-70.0	-13.0
B	#10	5	-61.8	- 1.8
C	#10	5	-20.0	+21.5

Difference (mm)(b-a)	μ^N/mm.
+57.0	19.0
+60.0	31.7
+41.5	16.2

-30-

DYNAMIC TESTS

Specimen	Gage Lt. (in.)	Reduced Diam. (in.)	Hammer Velocity (fps)
C	1.33	0.079	87
D	1.50	0.079	132
E	1.50	0.071	169

(a) The machine constructed delivered the
desired impact tensile load at velocities up
to 189 fps.

(b) Instrumentation used yielded a photographic
trace of strain in the specimen before and during
impact at 57 fps as shown in photograph (J).

(c) Elongation and reduction of area at various
velocities for 2S aluminum are noted below:

Specimen (see photo. J)	Velocity of loading (fps)	Nature of fracture
B	0 (static)	ragged concave
C	57	perfect concave
D	132	perfect concave
E	189	perfect concave

% Elongation	% R.A.
20	61.3
33	90.0
50	90.0
60	92.0

Sample Calculations:

Specimen 'B' (a) % elong. $= \dfrac{1.20-1.00}{1.00} \times 100 = 20\%$

(b) Initial Area = $\dfrac{(.262)}{4}$ = 0.02

Final Area = $\dfrac{(.109)}{4}$ = .0094

% RA = $\dfrac{0.05 - .0094}{0.05}$ × 100 = 81.3%

Photograph (J) is a print of the film
record obtained of impact at 57 fps. Traces
A, B and C are the zero positions of the forward
and rear gages mounted on the specimen and the
gage mounted on the holder, respectively.
Traces E and G indicate the final position of
A and B after fracture while the shutter was
still open. A faint trace can be seen of the
path of beam A from its original position to
position D. The wave superimposed upon traces
is 5000 cycle pick-up from the bridge-exciting
oscillator. The timing wave was off the film
to the right and would have appeared at the
right margin of the film if it had been realized
that the film did not extend to the entire width
of the viewing screen. Fortunately the 5000
cycle corresponds to .0002 sec. of elapsed time.
The distance from trace A to E represents 896
micro-inches/inch of strain and that from B to G
represents 1226 micro-inches/inch. The
significance of these figures is problematical.
They may represent open circuit values of

-34-

amplifier current. Or, if the bond between gage
and specimen failed before the circuit was broken,
they may represent residual strain in the gage
after the impact load was applied. At any rate
it can be seen that the gage was recording less
than 900 micro-inches of strain a full .001 of
a second after impact although at 57 fps the hammer is
traveling 0.68 inches in that time, or more than
the total elongation observed. It would seem
that either the gage is giving a false reading or
rupture is taking place at considerably below 57 fps.
The latter is the more likely explanation. A third
alternative is the possibility that galvanometer
response is not rapid enough to accurately trace
the signal. The inertia of the mirror, however
small, is finite and with an extremely rapid signal
it may tend to spread out the signal along the
time scale.

It is possible that too high a value of
amplification was used thus recording only the
first part of the actual trace before coming to the
limit of amplifier output. Decreasing gain would
sacrifice accuracy of readings, however, and this
is a factor to be considered.

Neither trace B nor C give a visible track
due possibly to a faulty rheostat on the
oscillograph which limited voltage that could
be applied to the lamp source of light. It
is not certain that trace C, the gage on the
steel holder, would have given any noticeable
reading under any condition.

The calculations given in Figure (9) and
shown graphically in Figure (10) result from
an application of the von Karman theory to the
calculation of critical velocity for the 2S
aluminum. This value is about 35 fps. According
to the theory summarized earlier, at impact
velocities greater than 35 fps, fracture should
be characterized by small total elongation.
Figures (11) and (12) show that experimental
data obtained at higher hammer velocities
shows greater elongation than under static con-
ditions. This indicates again the possibility that
a considerable difference may exist between hammer
velocity upon impact and true velocity of the
end of the sample. It is obvious that the velocity
of the end of the sample must accelerate from
zero to an undetermined value at fracture so that

-86-

the purely theoretical case of instantaneous
attainment of velocity can only be approached,
and analyses of results must be made with this
in mind. The exact relationship of hammer
velocity and sample velocity is the unknown here.
Much more work is needed to clear this up.

Also of interest is the location of the
point of fracture on the sample tested at 139 fps.
(Specimen 3, Photo I). The fact that it broke
away from the moving end and has strong evidence
of double-necking might possibly be explained
by the reflection and super-position of stress
waves. The double-necking phenomena has been
reported in previous work.[1]

X   CONCLUSIONS

The authors are aware that definite conclusions cannot be based upon the small number of tests that have been performed. However, the results of this investigation seem to indicate that the following facts may be used as a basis for further investigations.

1. The basic principle of the loading machine used in this investigation is sound and may be satisfactorily applied to impact testing at velocities up to at least 200 feet per second.

2. The use of an electrical resistance type strain gage, together with a General Electric Type FM-10-92 Oscillograph for recording strain during impact is probably limited to velocities under 100 feet per second.

3. Aluminum in the 2S form is more ductile when loaded at hammer velocities up to 109 fps than it is under static conditions.

Suggestions for future work in the field
of high velocity impact are divided into two
groups, namely, one group which includes
suggestions pertaining to modifications of the
testing procedure and apparatus used in this
investigation, and another group which relate
in a general way to further experimental projects
in the subject.

A. Suggestions for improvement of the existing
apparatus are as follows:

1. Make the ring, hammer, pin, tripping plunger
and tup of the highest grade tool steel available
and make certain that the finished product is
heat treated to yield the proper balance between
high strength and toughness.

2. Redesign the hammer, pin and ring to provide
for the terrific loads imparted to them. Detailed
design of these parts as shown in Figure (6) is
recommended for trial.

3. Place tripping plunger assembly as far forward
as possible to avoid tripping hammer too soon.

4. Provide lateral bracing for the bearing support

on the side to which the tripping plunger assembly
is attached. This bracing will carry the thrust
caused by the tripping ring as it turns about
the tube.

    4. Shorten instrument leads to minimum length
and ground at both ends, to eliminate pick up.
Possibly isolated oscillators may improve situation.

    5. Devise an electrical trip for the hammer
actuated by the oscillograph transient control
switch to synchronize automatic camera exposure
with impact, or use continuous record film.

    7. Handle strain gage leads with extreme care
to eliminate grounding on sample holder or
interference with hammer. Scotch tape will hold
leads in place.

    8. A small D.C. generator mounted on one end
of the main shaft of the impact machine could be
used to give a continuous record of rotating
speed on the film if properly calibrated. This
might be useful in investigating hammer-tup speed
relationship, or in determining energy values.

    9. Use 8 inch width film for recording to allow
maximum amplitude for signals.

10. Consider extending length of sample
by moving stationary holder away from tup.
Would make it possible to use larger gage
loading area, would facilitate mounting specimen
by making it unnecessary to pass gages through
slot in holder, and would make it easier to
detect wave phenomena by increasing distance
between gages.

11. Mark a grid upon reduced section of sample
and measure variations in spacing after impact
to obtain better overall picture of plastic flow
in different parts of the section.

12. Use timing curve of a frequency to give 20
cycles per inch of record film.

13. Consider stepping up pulley ratio from
camera motor to camera to obtain speed greater
than 1270 ft. per minute with magazine film.

14. Use super-sensitive film to obtain more
positive track.

15. Use following settings on instruments:
(a) Attenuator setting 6 or 5 on amplifier to keep
strain record within limits of film. On steel
mounted gage use maximum gain.

(b) Equilibrium current less than 40 ma to
avoid signal saturation by amplifier.

(c) Resistance setting 6 Ω on galvanometer to
decrease pick-up.

(d) Voltage about 10 V or less during exposure
to insure visible track.

B. FUTURE WORK

1. Consider a rotating device with a fixed
hammer that clears the fixed specimen until a movable
tan springs into position to cause the hammer to
engage the specimen. This was used as described
in reference (10).

2. Try cathode ray oscilloscope to obtain most
sensitive response and use high speed motion
picture camera for recording record.

3. Record gage signals on magnetic tape or wire
recorder and play back at reduced speed for film
record.

4. Experiment with brittle lacquer, determine its qualitative
possibilities in the region of brittle fractures.

5. Use high speed motion picture camera on specimen
during impact to determine mechanism of fracture.

6. Devise mechanism to record velocity of tup versus
time during fracture. Might be done by using coil on
tup and a strong magnetic field around it, or by
electro-magnetic gage to indicate displacement
versus time. This would give a positive record of
true velocity of end of specimen.

XII SKETCHES, DIAGRAMS
and PHOTOGRAPHS
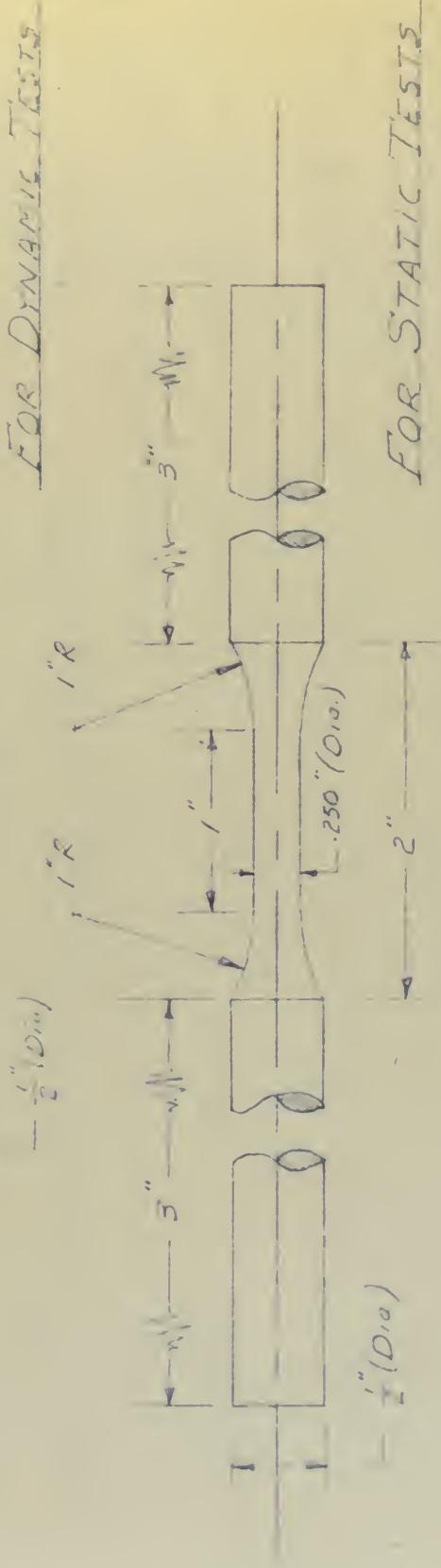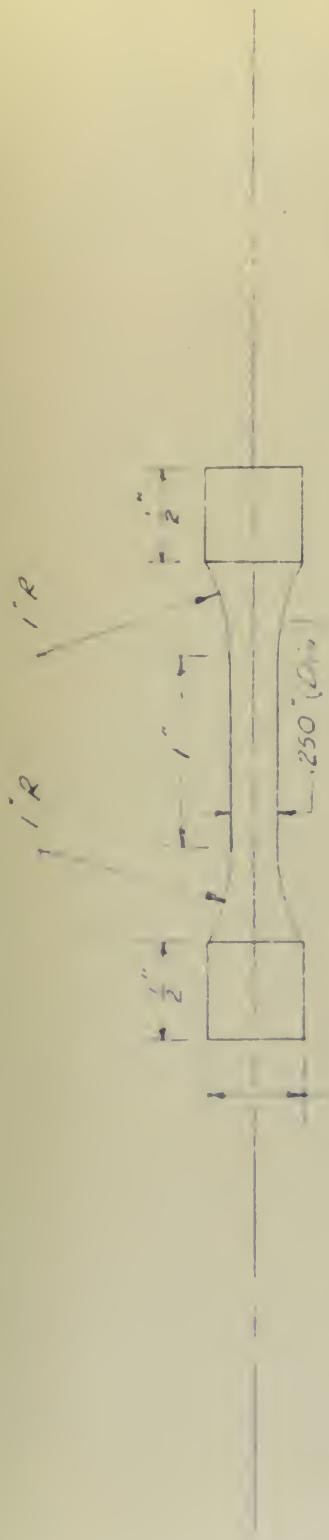
FOR DYNAMIC TESTS
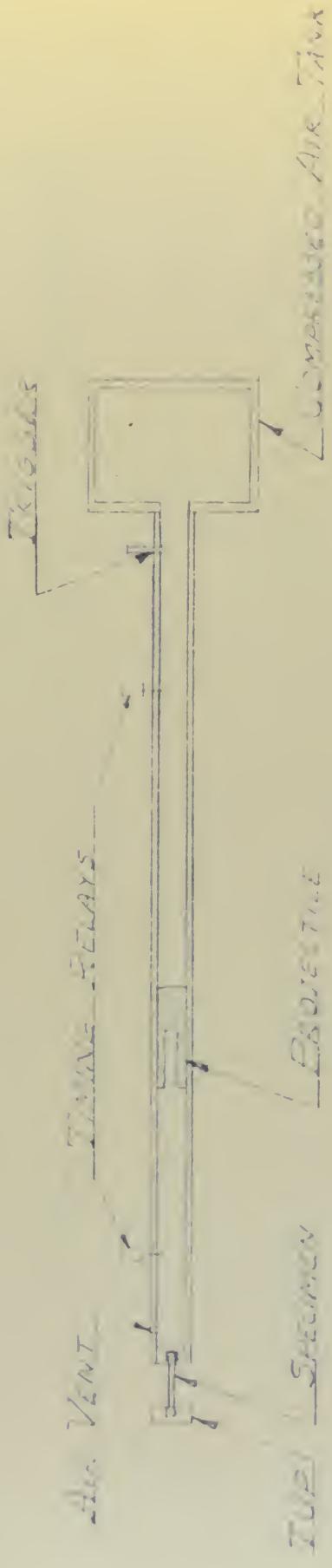
FOR STATIC TESTS

FIG. 1

SPECIMENS

Fig. 2

A Scheme For Impact Loading

BALL
PROJECTILE   GATE TUBE   SPECIMEN

WHEEL

SHAFT   WHEEL

DRUM   DRUM

                                        SECTION A·A

FIG 3

A SCHEME FOR IMPACT LOADING

SHAFT

FORKED HAMMER

SPECIMEN

TUP

HAMMER

TUP

FIG 4.

A SCHEME FOR IMPACT LOADING

HAMMER & PIN - FULL SCALE

PIN
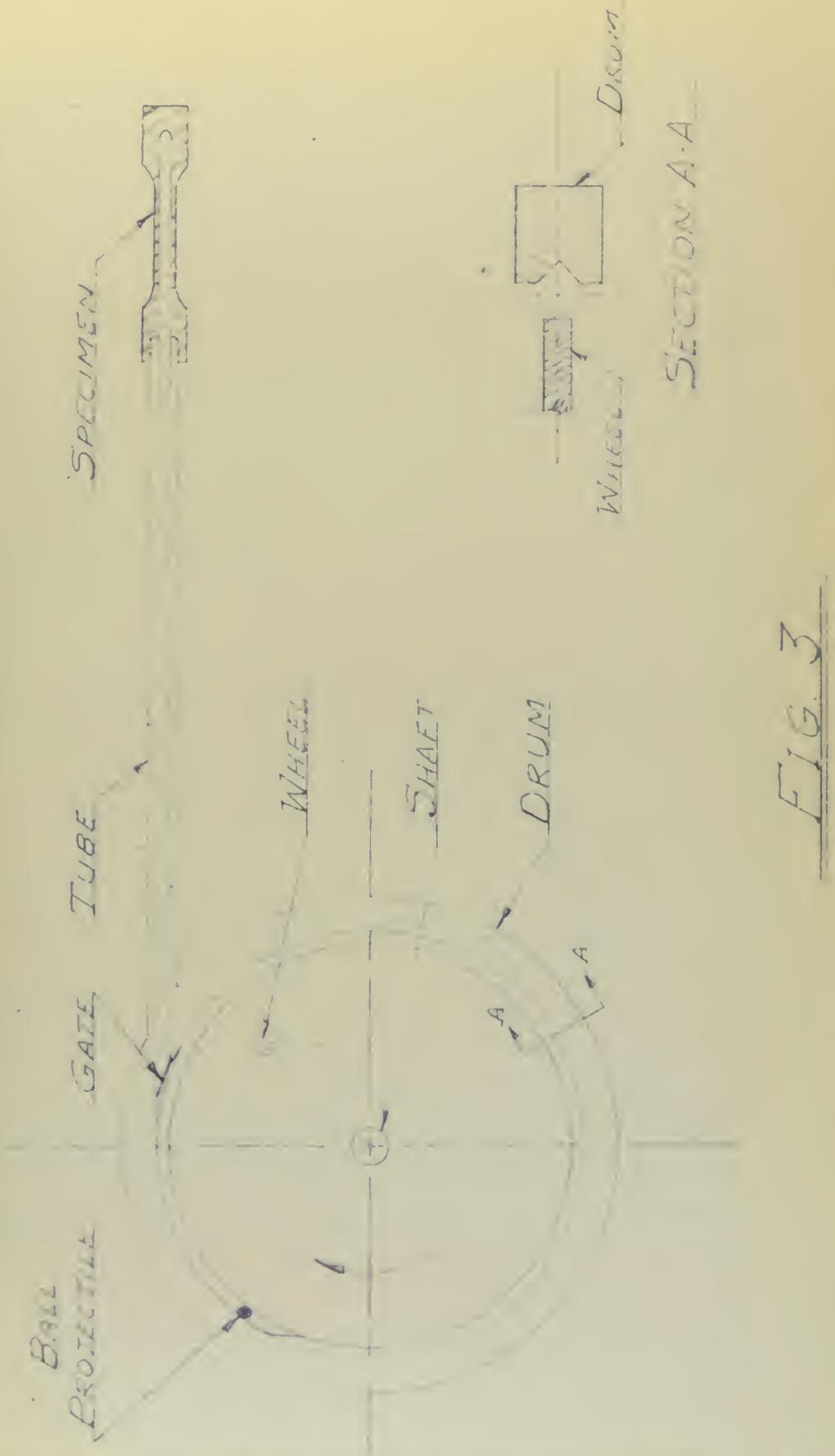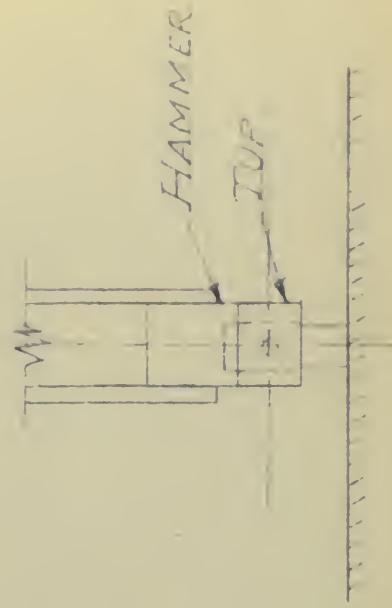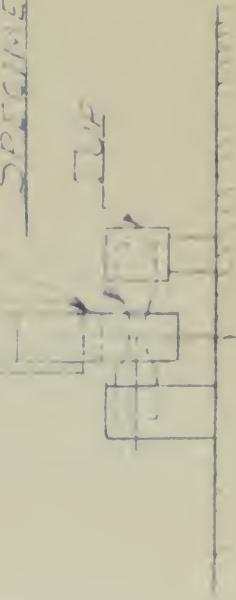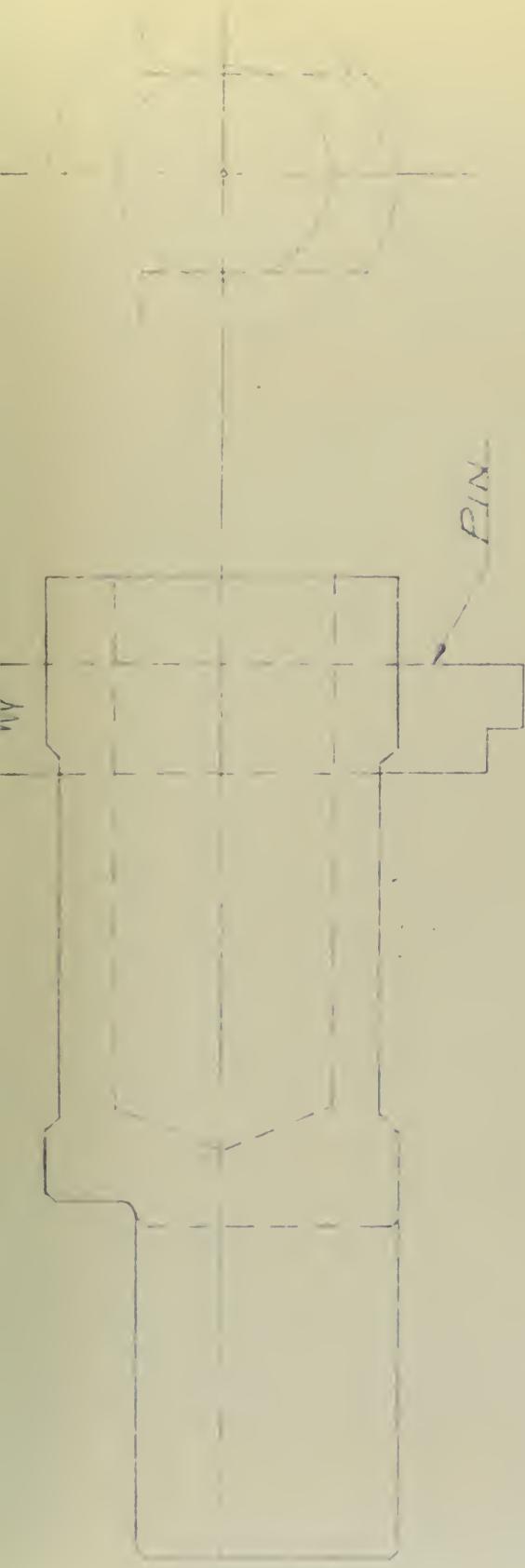
RING - HALF SCALE

SLOT

FIG. 5

DETAILS OF LOADING MACHINE
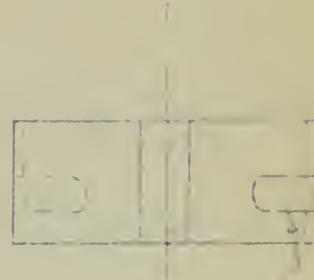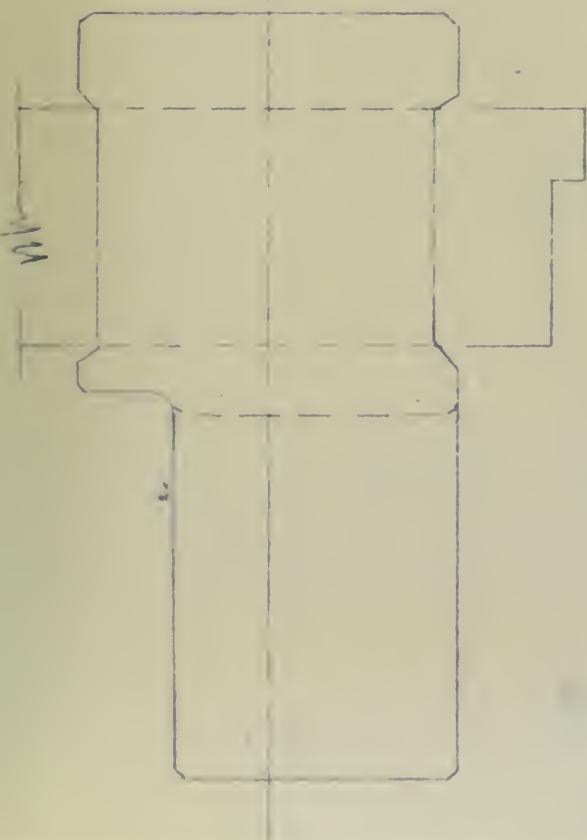
HAMMER & PIN - FULL SCALE

RING - HALF SCALE

FIG. 6

PROPOSED MODIFICATION
OF LOADING MACHINE
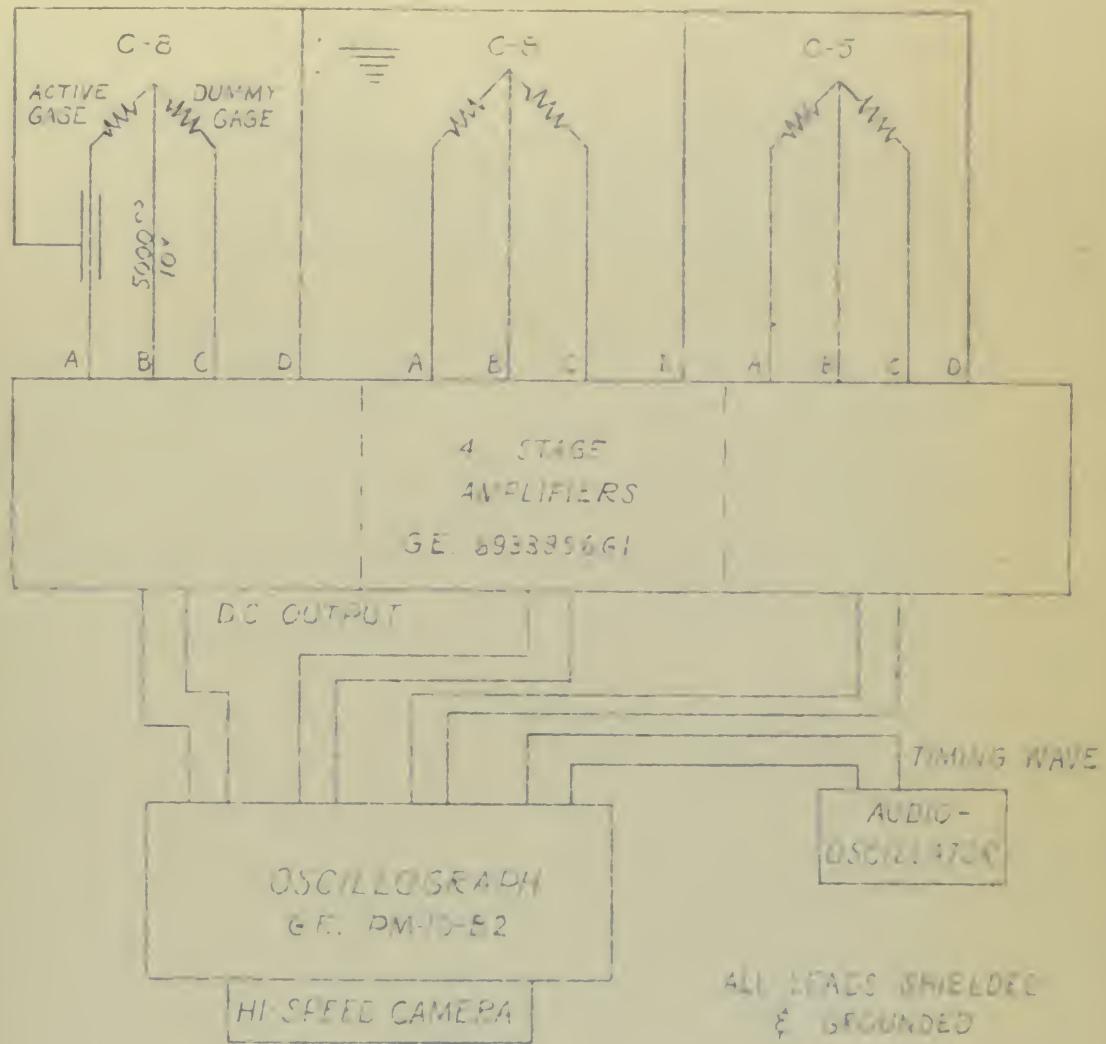
FIGURE 7 - INSTRUMENTATION

FIGURE 8

STATIC STRESS-STRAIN CURVE

2S - ALUMINUM

FRACTURE

WELD
STRENGTH

$E = \dfrac{8192}{.00087} = 9,350,050$ PSI

YIELD AT .002" SET 12,000 PSI

ULTIMATE = 15,300 PSI

ELONGATION = 20% IN 1/4

RA = 81.3 %

σ
(PSI)

ε strain, in/in

FIGURE #9

V_ε vs ε Curve

Calculations

$$V_\varepsilon = \sqrt{\frac{d\sigma}{d\varepsilon}/\rho} \quad \text{Where } \rho = 160 \frac{lb}{ft^3} \cdot \frac{sec^2}{32.2 ft} = 4.97 \frac{lb \, sec^2}{ft^4}$$

$$V_\varepsilon = \frac{1}{2.23} \sqrt{\frac{d\sigma}{d\varepsilon}}$$

From ε	To ε	$\Delta\varepsilon$	$\frac{d\sigma}{d\varepsilon}$ lb/in^2	$\sqrt{\frac{d\sigma}{d\varepsilon}}$	V_ε	$V_\varepsilon \cdot \Delta\varepsilon$
0	900×10^{-6}	900×10^{-6}	1343×10^6	36.7×10^3	16.4×10^3	14.81
900×10^{-6}	1000	100	1159	33.8	15.1	1.51
1000	1100	100	1080	32.9	14.7	1.47
1100	1200	100	1008	31.8	14.3	1.43
1200	1300	100	922	30.4	13.6	1.36
1300	1400	100	879	29.7	13.3	1.33
1400	1500	100	792	28.2	12.6	1.26
1500	1600	100	691	26.3	11.8	1.18
1600	1700	100	663	25.8	11.5	1.15
1700	1800	100	590	24.3	10.9	1.09
1800	1900	100	533	23.1	10.3	1.03
1900	2000	100	461	21.5	9.6	0.96
2000	2100	100	374	19.3	8.7	0.87
2100	2200	100	346	18.6	8.3	0.83
2200	2300	100	302	17.4	7.8	0.78
2300	2400	100	245	15.8	7.0	0.70
2400	2500	100	216	14.7	6.6	0.66
2500	2600	100	196	14.0	6.3	1.33
2600	-	-	neglig.	-	-	-

$$\sum = 34.31$$

Critical Velocity = 34.3 fps
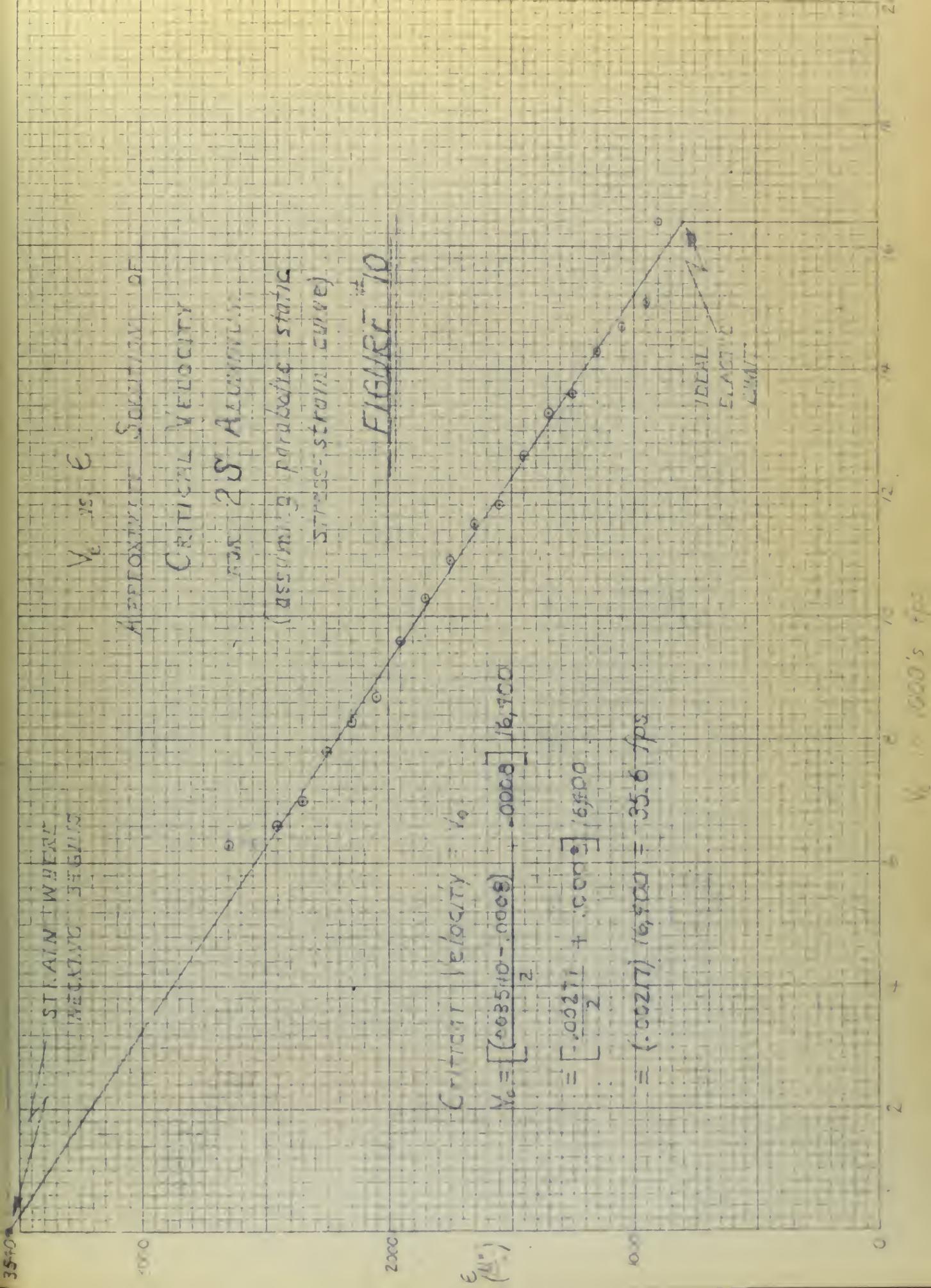
STRAIN TWEER METHOD, STATIC FIGURE

V_c vs ϵ

APPROXIMATE SOLUTION OF

CRITICAL VELOCITY

FOR 2S ALUMINUM

(assuming parabolic static
stress-strain curve)

FIGURE 10

Critical Velocity $= V_c$

$$V_c = \left[\frac{(.003540 - .0008)}{2} - .0008\right] 16,900$$

$$= \left[\frac{.002741}{2} + .0008\right] 16,900$$

$$= (.00277) 16,900 = 35.6 \text{ fps}$$

ϵ $\left(\frac{in.}{in.}\right)$

3540

3000

2000

1000

IDEAL ELASTIC LAW

V_c - 000's fps

FIGURE #11
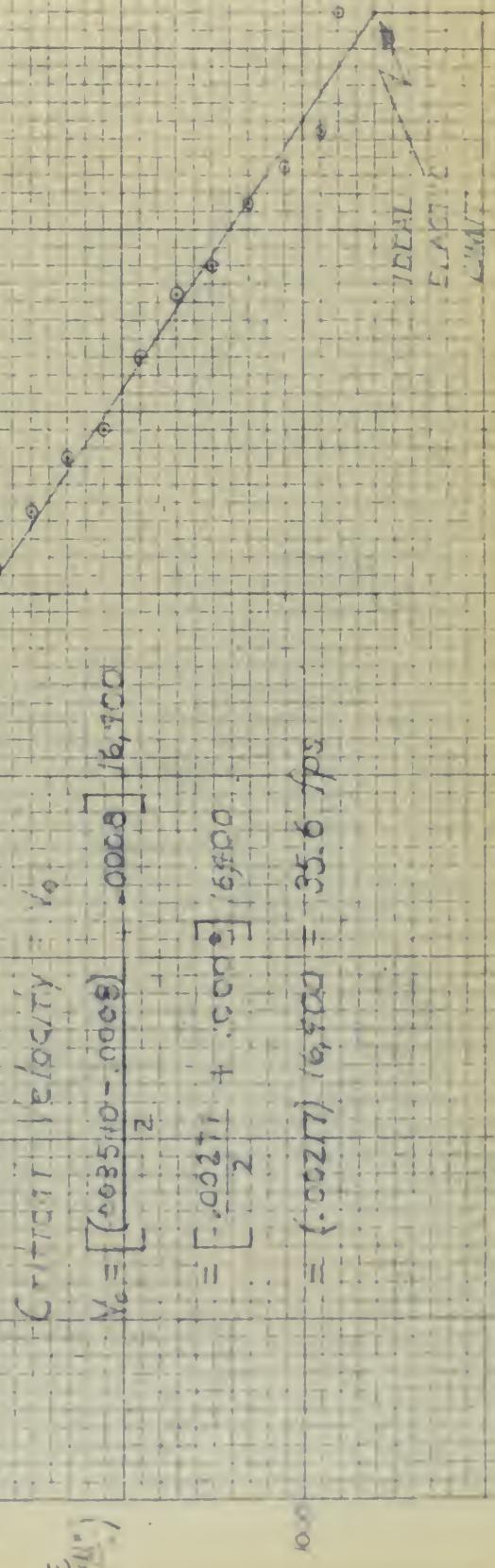
ELONGATION "A-1" vs.
IMPACT VELOCITY.
2S ALUMINUM.

% ELONGATION

IMPACT VELOCITY (fps)

FIGURE 12

REDUCTION IN AREA
vs
IMPACT VELOCITY

(.25 φ SPECIMEN)
28% ALUMINUM
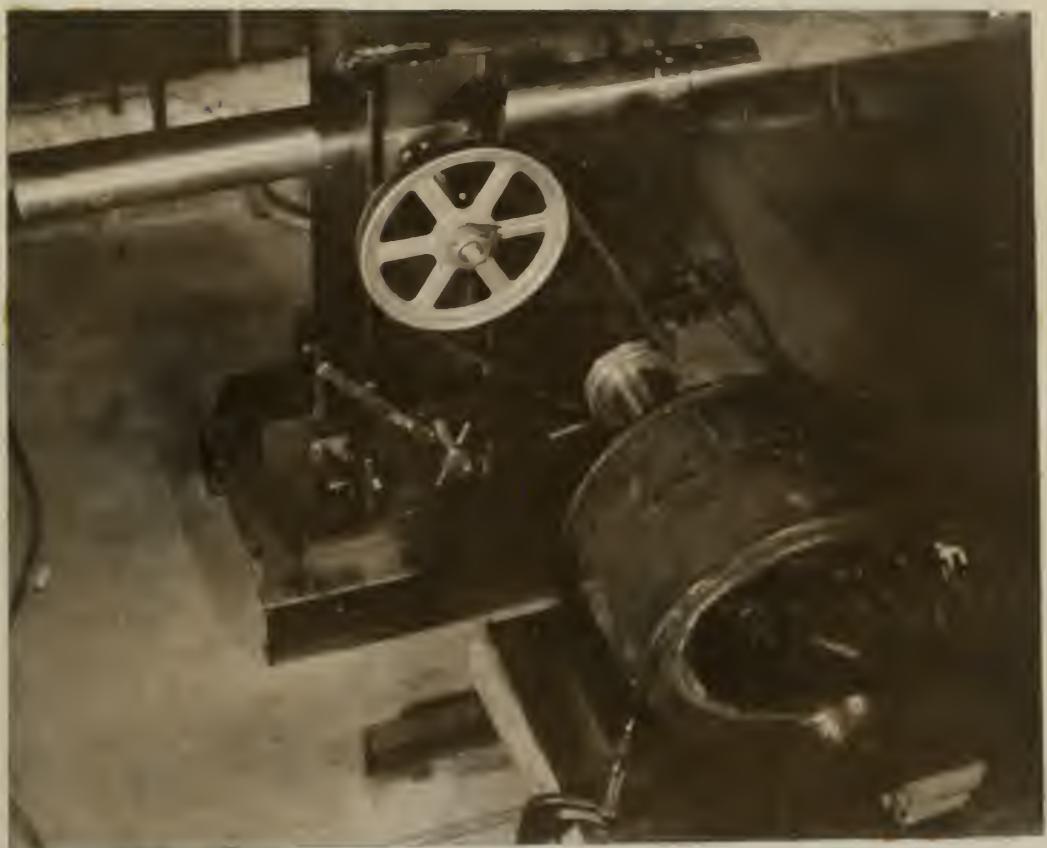
PHOTO A
General view of Loading Machine
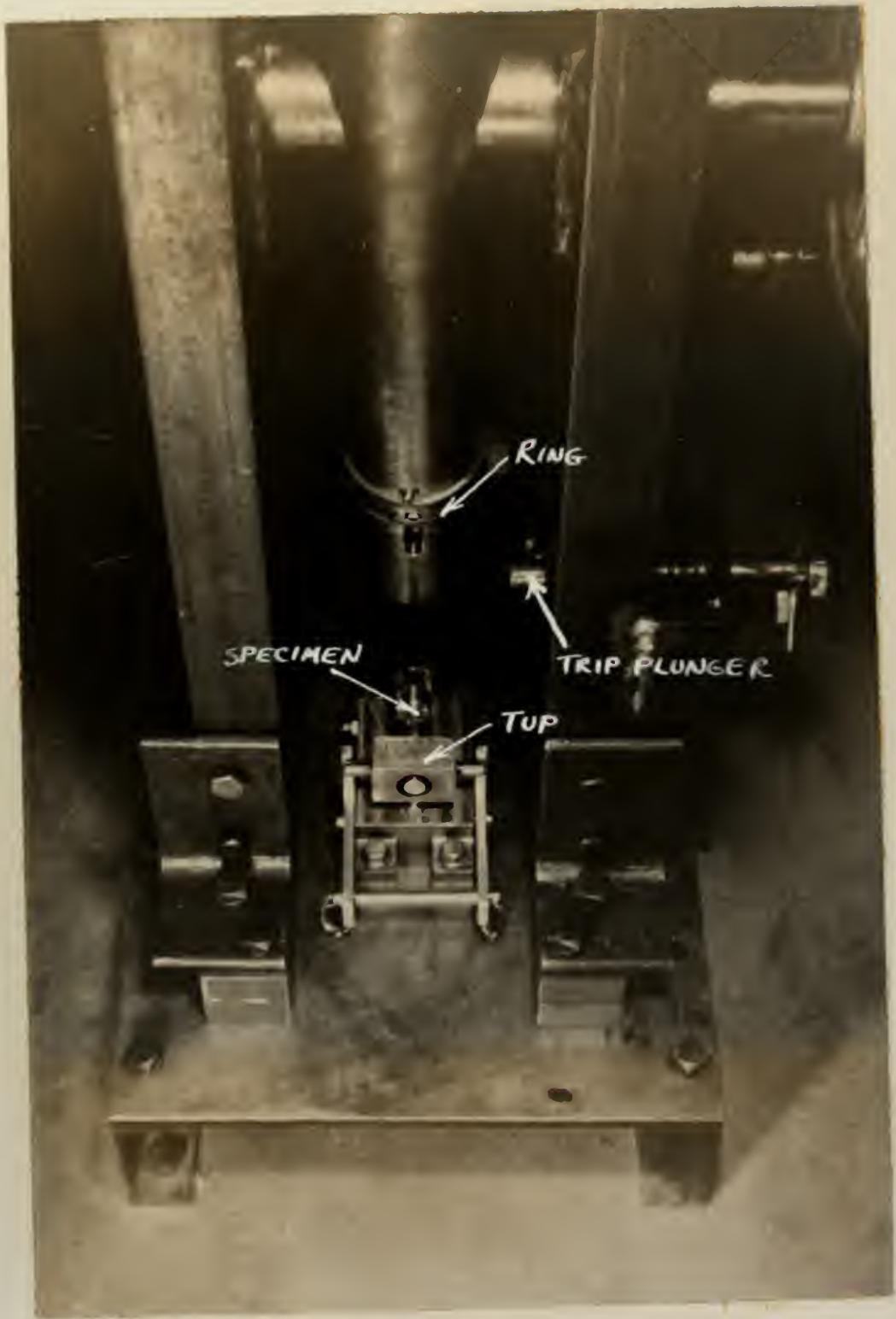
PHOTO B
Front view of Loading Machine
showing tup, tube and hammer
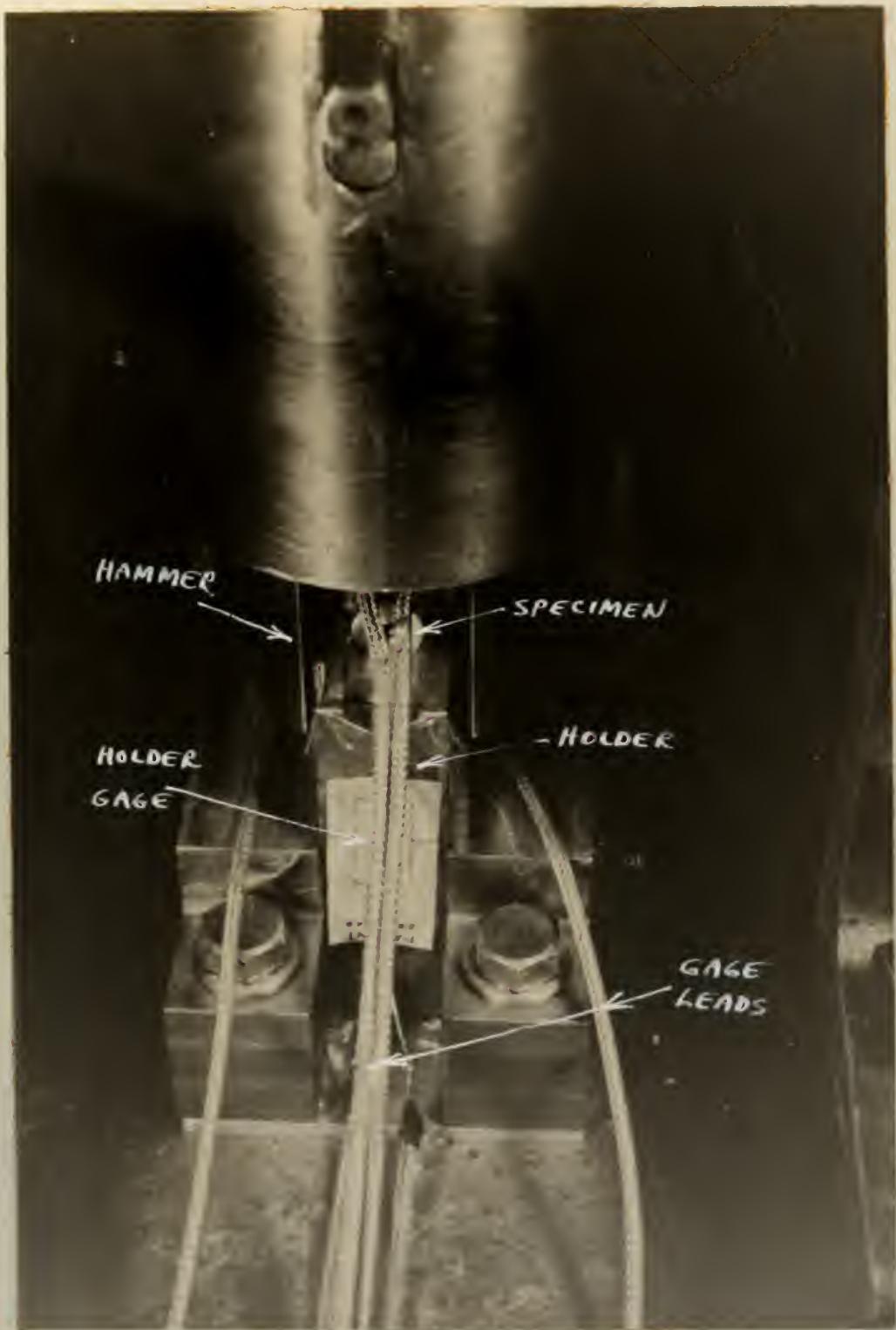
PHOTO C
Rear view of Loading Machine
showing tube, hammer, holder and gages
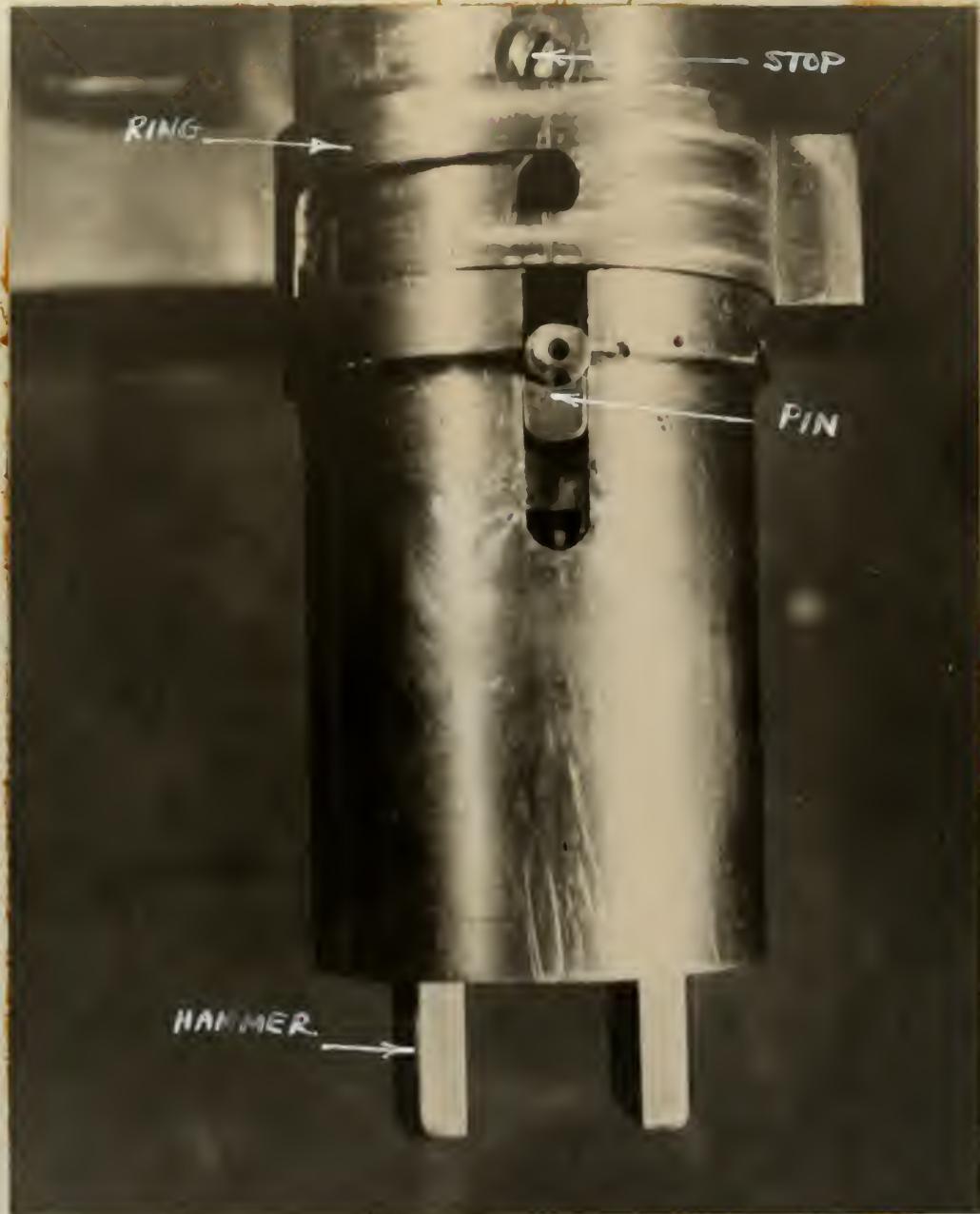
PHOTO D
View of Hammer and Ring
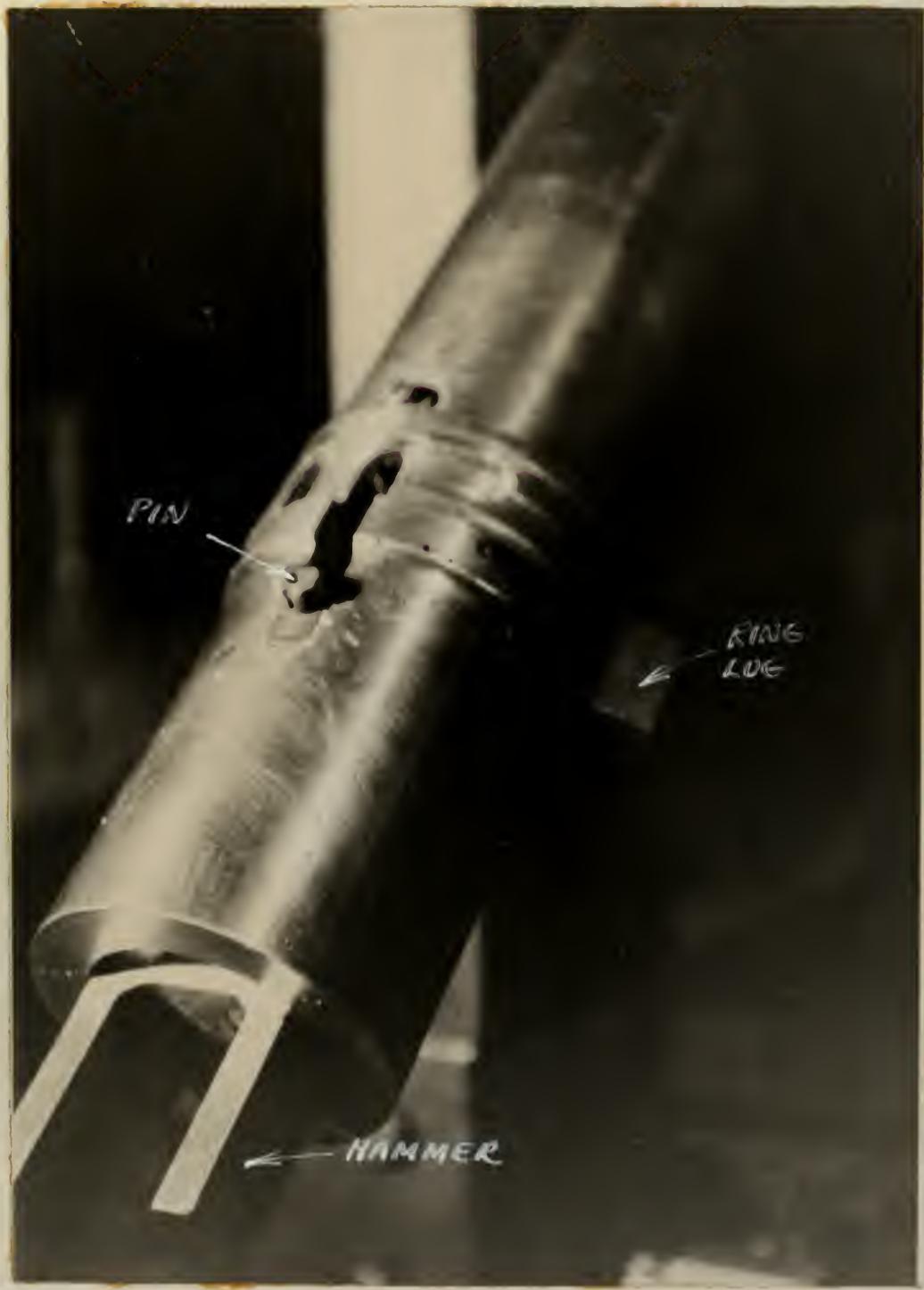
PHOTO E
View of Hammer and Ring
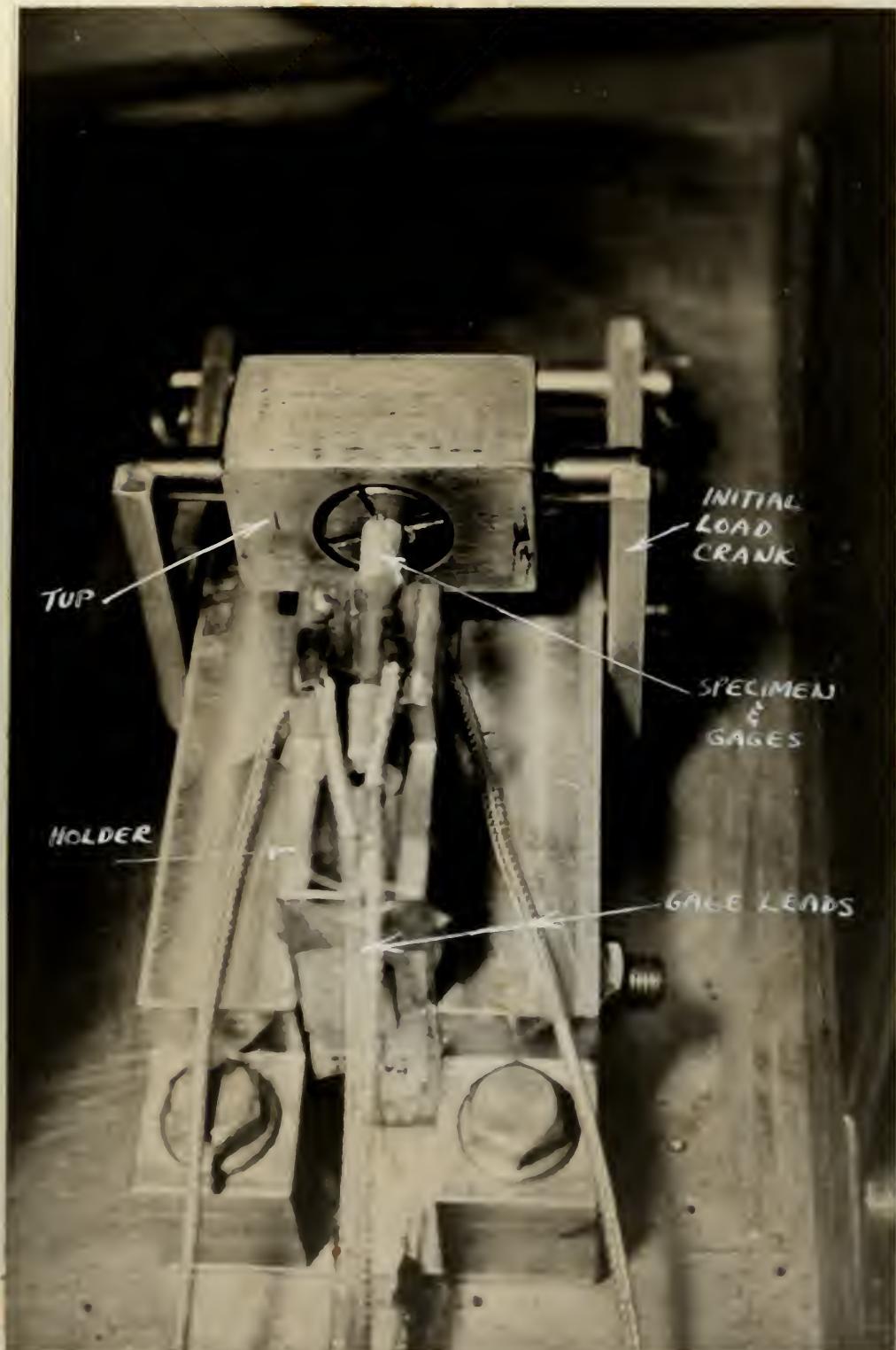
PHOTO F.
View of Tup, Holder and Specimen
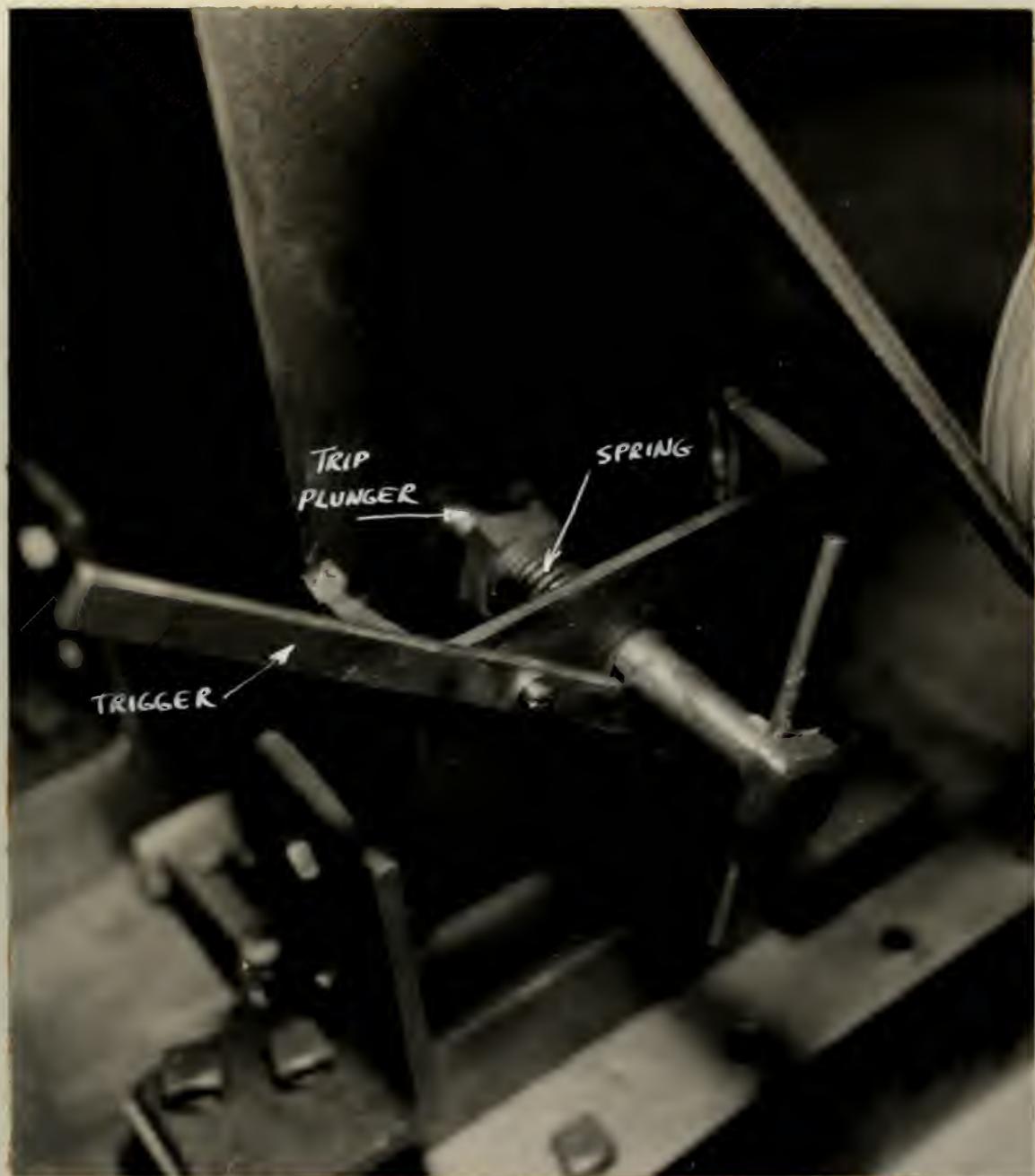
PHOTO G
View of Tripping Plunger

PHOTO B
View of Instruments
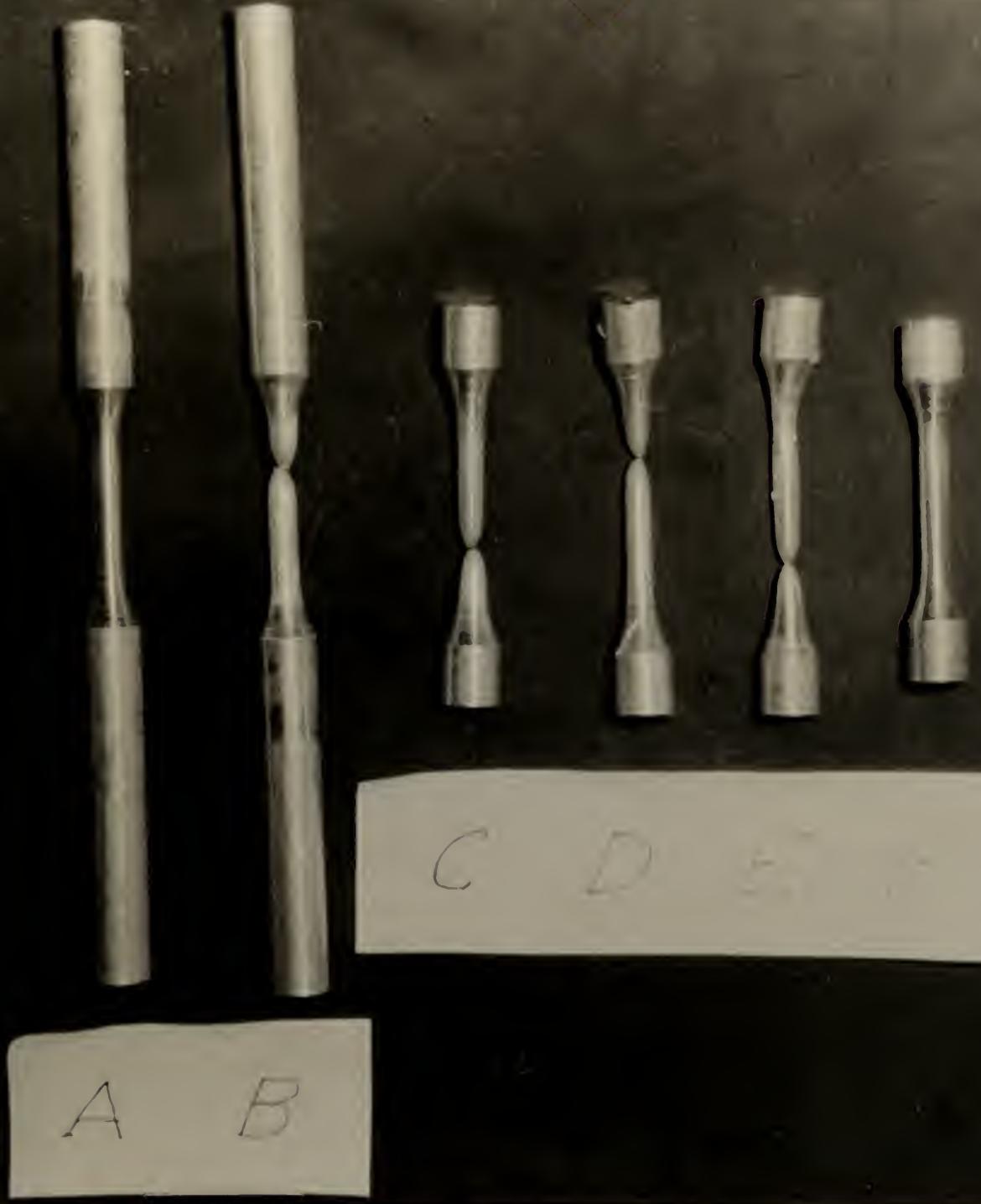
PHOTO I

Specimen A - Static Before Testing
"   B -   "   After   "
"   C - Impact - Fractured at 57 fps
"   D -   "   -   "   " 132 "
"   E -   "   -   "   " 189 "
"   F -   "   - Before testing

TIME

FRACTURE
TRACES

A   B   D   C E

PHOTO J
Oscillographic Record
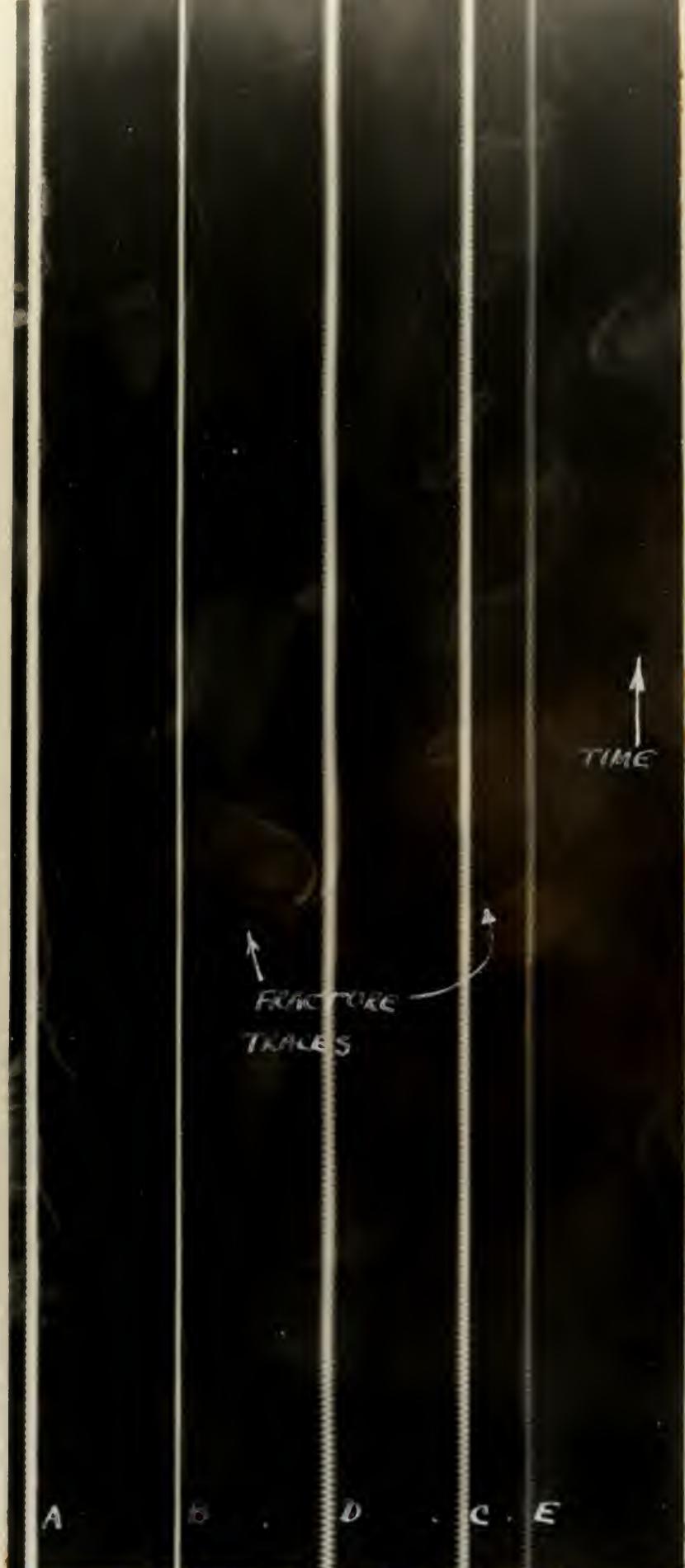
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DATE DUE			